# Evaluation of rotor vibration in gearboxes

*by Steven M. Schultheis, P.E.*
*Principal Engineer*
*Machinery Diagnostic Services*
*Bently Nevada Corporation*
*stschu@aol.com*

Much has been written over the years regarding vibration analysis of gearboxes, but the primary focus has been on the unusual characteristics exhibited, due to the interaction of the gears. Malfunctions related to gear mesh and tooth combination, as well as manufacturing problems, such as ghost frequencies, are typical topics of discussion in gear presentations. This article has a somewhat different focus. Gear shafts exhibit rotor behavior that is very different from a turbine or a compressor. Their unique behavior makes gearboxes an interesting piece of equipment to evaluate. This article will focus on gearboxes with fluid film bearings instrumented with proximity probe transducers, and the evaluation of the shaft vibration data.

In a turbine or compressor with fluid film bearings, one of the primary forces acting on the shaft is gravity. In general, it is the gravity load of the rotor in the bearing that the bearing is designed to support. In gearboxes, gravity is still a load, but it is overshadowed by two other forces: the gear separation force and the tangential force due to transmission of torque through the gear mesh. As a result, the net load on the bearing is not necessarily vertically downward as in other machines. In the upmesh-pinion, speed-increasing gearset (Figure 1), assuming a lightweight pinion, the net load is up and to the left on the pinion, and down and to the right on the gear. If the pinion weight exceeds the tangential force, then the net load on the pinion is down and to the left. These forces need to be accounted for in the evaluation of shaft centerline position plots and orbit plots of gear shafts. This is also the reason that, on some gearboxes, proximity probes are not mounted at 45°L and 45°R. It is quite common for the probes to be mounted at 75°

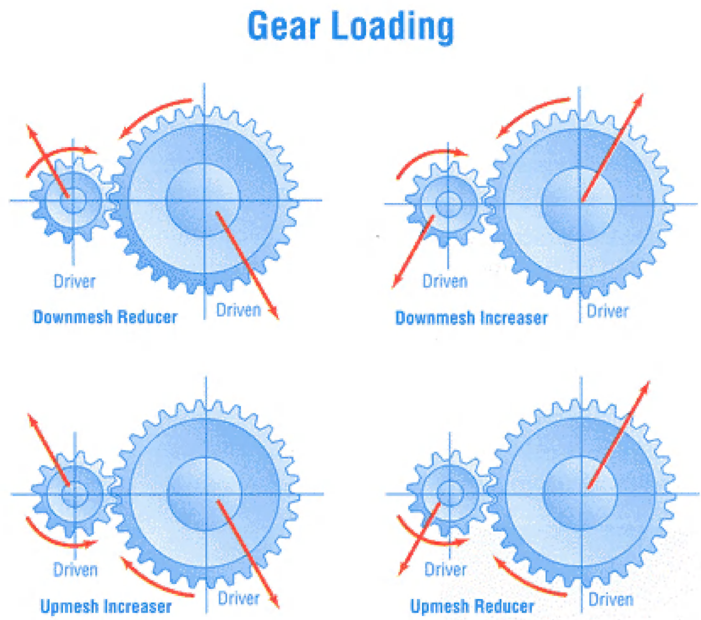


Figure 1. Gear forces on a gearset

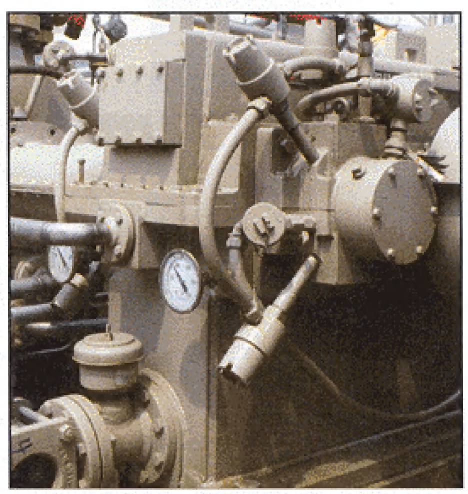

**Figure 2. Probe installation on a pinion, 45°L and 135°L.**

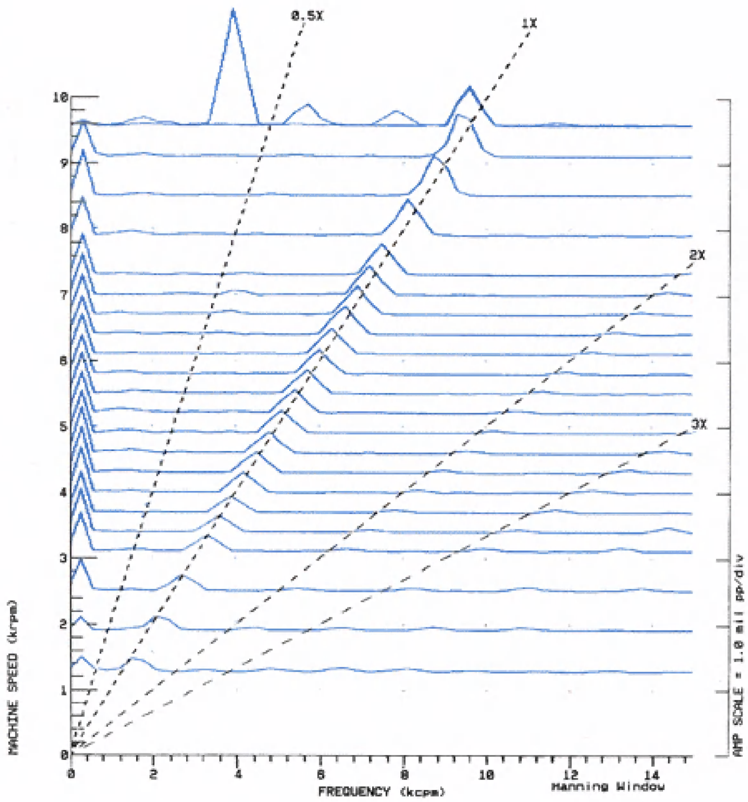


**Figure 3. Spectrum cascade plot, pinion coupling-end startup data.**

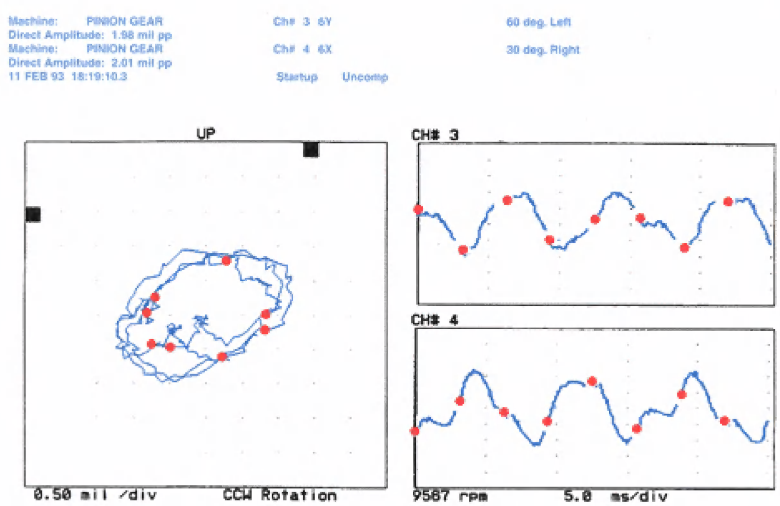


**Figure 4. Orbit/timebase plot at the pinion coupling-end, no load on the compressor.**

left and 15° right on the pinion, and 15° left, 75° right on the bull gear. This may be done to facilitate installation, but it also places the probes more in line with the major and minor axes of the orbit for the gears. As is shown in Figure 2, probes may also be installed at other angles to facilitate the installation. In this case, the probes for the pinion are 45°L and 135°L.

The following case histories demonstrate some of the unique characteristics of gear shaft behavior.

### Case history 1, pinion loading

One of the most common machine train configurations in chemical plant and refinery service is the motor-gearbox-compressor train. In these trains, the motor typically runs at 1800 or 3600 rpm, and the gearbox increases the speed for the compressor. In these trains, it is also very common to use an upmesh pinion. In one configuration, as is shown at the bottom left of Figure 1, the bull gear turns clockwise and the pinion turns counterclockwise, so that the net force on the pinion is up and to the left. Under full speed and load conditions, the pinion should be heavily loaded in the upper left quadrant of the bearing. As the machine is starting up, though, the torque load, and thus the gear forces, has not fully developed, and the machine cases are not thermally stable. There is a range of speeds and alignment conditions where the net gear load can cancel out the gravity load. As a result, the pinion exerts nearly zero load on the bearing.

The most common result is fluid-induced instability, commonly referred to as oil whirl. As the shaft eccentricity in the bearing approaches zero, the Fluid Circumferential Average Velocity Ratio λ increases to approximately .43, and the Dynamic Stiffness decreases. This causes the Threshold of Stability frequency to

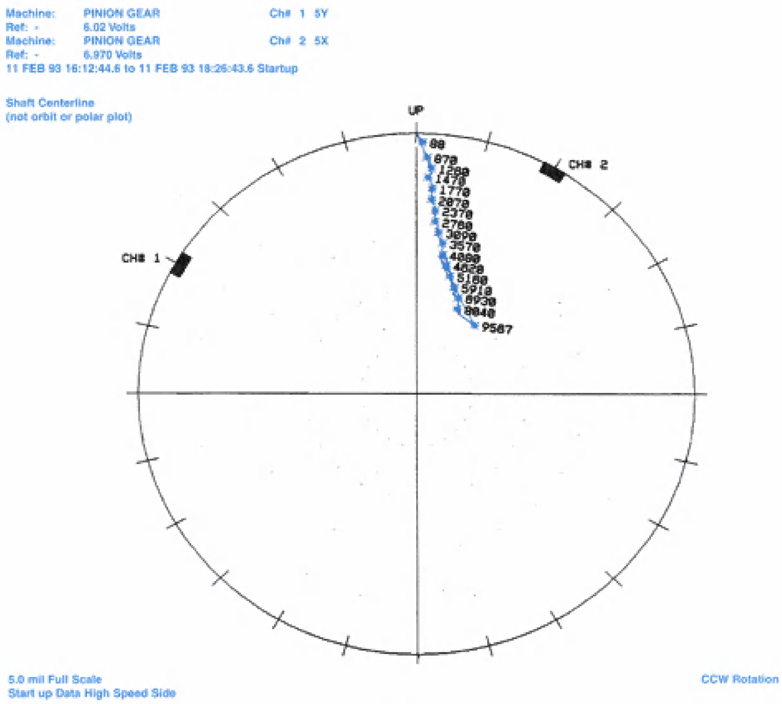


**Figure 5. Shaft centerline plot, pinion coupling-end, during startup.**

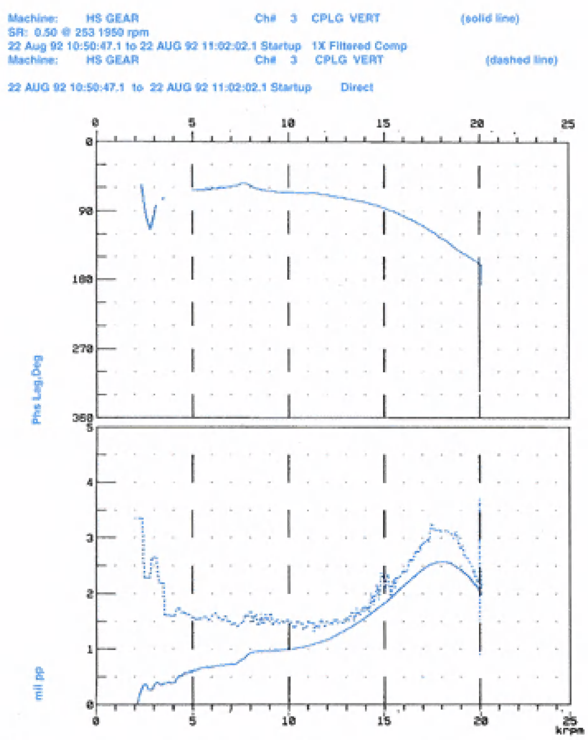


**Figure 6. Bode plot of the coupling-end bearing vibration response. The pinion passes through resonance at approximately 19,000 rpm.**

decrease. When this frequency equals the running speed, the fluid-induced instability occurs, and the pinion shaft vibrates at .43X (43% of running speed). One of the primary methods to correct a fluid-induced instability is to increase shaft eccentricity ratio to nearly 1. This is easily accomplished in a gearset by loading up the driven machine. Increasing the load increases both the gear separation force and the tangential force, moving the pinion to the top left quadrant of the bearing. As the pinion load increases, the eccentricity and bearing stiffness increase and $\lambda$ decreases, and oil whirl stops. Figures 3 and 4 show spectrum cascade and orbit/timebase data from the pinion during a startup of a gearset in a wet gas compressor train where the pinion experiences an instability. The instability begins just below the 9567 rpm running speed of the machine.

The shaft centerline plot (Figure 5) for this same startup shows that, due to alignment offsets, the pinion actually started in the top of the bearing. As the gearbox approached running speed, the average position of the pinion moved to near the center of the bearing. As a result, the pinion went unstable. The solution to this problem was to load the compressor. Loading the compressor increased the gear forces which acted to increase shaft eccentricity and stabilize the pinion.

The wet gas compressor train is intended to be started only every few years, so it is possible to live with this transient instability problem. For a unit that is started on a regular basis, the problem could be addressed with a different bearing design.

### Case history 2, resonance

One general rule of thumb for gearsets is that, since gear shafts tend to be very short and stiff, they typically run well below their first

balance resonance. In general, a Bode plot for a gear element will show, with increasing speed, a constant phase lag angle and a gently increasing amplitude. The Bode plot in Figure 6 shows the 1X response for a high speed pinion in a gearset with an input speed of 3600 rpm and an output of 19,865 rpm. The Bode plot indicates that the pinion is running just slightly above a balance resonance. From the frequency of the amplitude peak and 90° phase shift, we would conclude that the resonance is at approximately 19,000 rpm. This is unusual and, based on the vibration level of 64 to 76 μm pp (2.5 to 3 mil pp), not at all satisfactory. The problem in this case was due to a very long shaft extension on the pinion, and a very heavy coupling, which lowered the resonance frequency of the pinion. The problem was solved by shortening the pinion shaft extension and using a reduced-moment coupling, although it involved resetting the spacing between the machine cases. The vibration characteristics upon restart were quite satisfactory.

### Case history 3, thermal growth

Interpretation of shaft centerline data relies on having a valid gap reference. On a turbine, this is normally accomplished by sampling the gap voltage with the machine shut down. At this point, it is safe to assume that the shaft is resting in the bottom of the bearings. For gearsets, this is not always a safe assumption. For very light pinions, less than 222 N (50 lb), the pinion could be hung on the bull gear. Also, since pinions can be very light, alignment offsets between the pinion and the driven machine may allow the coupling to hold the pinion off the bottom of its bearing. Figure 7 shows another shaft centerline plot for the coupling end of a pinion. This plot shows how difficult it can be to select a reference voltage. At slow roll speed, the pinion appears to sit in the bottom of

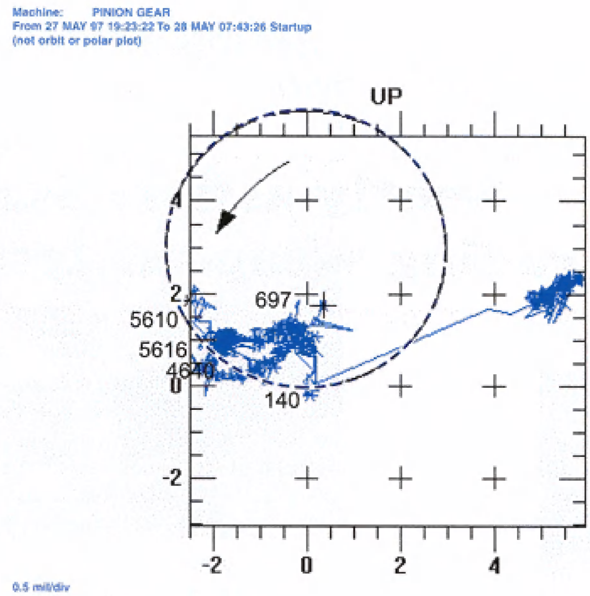


**Figure 7. Shaft centerline plot for the coupling end of the pinion.**

the bearing. Gap voltage data, acquired as the unit cooled down overnight, showed that the pinion moved up approximately 25 μm (1 mil), and to the right 102 μm (4 mil). This is most likely due to the alignment offset between the gear and the driven machine, and may indicate greater clearance than was expected.

A complicating factor would exist if the maintenance crew had used a thick sealant, such as silicone RTV, to seal the splitline of the gearbox during the turnaround. Since the bearings are retained by the gearbox cover, if the splitline is held open with the sealant, then the bearings can move relative to the case. Since the proximity probes are typically mounted to the gearbox lid, both the shaft and the bearing can now move relative to the proximity probes. This type of condition could also explain the excessive shaft centerline movement. In general do not use an RTV sealant compound that would affect a controlled dimension for sealing machine components, for example, radial or thrust bearing housings.

### Conclusions

Evaluation of shaft vibration and position on a gearbox poses some interesting challenges. Like any other type of equipment though, understanding the fundamental dynamics of the machine helps in evaluating these characteristics. Applying the experience gained in evaluating compressors or turbines, however, can lead to incorrect conclusions with gearboxes. The basic assumption that the net load on the shaft is vertically downward, due to gravity, may not apply. Understanding the gear forces helps you understand the shaft centerline position changes and other behaviors common to gear elements.

### Reference


1. Winterton, J., "A Tutorial on Gearing - covering certain aspects of design & dynamics," Bently Nevada Corporation.